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R81-30
12 March 1981

AD A098808

TECHNICAL REPORT

STABILIZATION FEASIBILITY FOR AN UNBALANCED 90mm GUN

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Prepared for
U.S. ARMY
ARMAMENT RESEARCH AND DEVELOPMENT COMMAND
Dover, New Jersey 07801

In response to
Contract DAAK 10-78-C-0313
Modification P00008

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Delco Electronics

General Motors Corporation
- Santa Barbara Operations
Santa Barbara, California

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TABLE OF CONTENTS

ILLUSTRATIONS

TABLES

SECTION I
INTRODUCTION

The objective of this study was to determine the feasibility of stabilizing an ARES 90mm gun on a 22-ton class combat vehicle. Since the major difference between the ARES gun and other guns currently on combat vehicles (from a stabilization point of view) is the large unbalance, this study is primarily concerned with the effect of the unbalance.

The primary phenomenon of concern is acceleration's act on an unbalanced gun to produce torques in the drive system that move the gun away from its desired position. For turret traverse drives, these torques are small and do not usually constitute a significant error source. However, the impact on an elevation drive system can be large enough to produce unacceptable gun positioning errors. The magnitude of these errors is a function of the disturbances, the amount of unbalance, and the characteristics of the components in the drive system.

In this report, the relationship of gun servo performance to hardware parameters is described in a qualitative and quantitative manner. The reasons for selecting a specific gun drive system are described, and a mechanization is defined for a servo to position the ARES 90mm gun. This servo is then analyzed to determine the pointing accuracy that can be expected.

SECTION II
SUMMARY AND CONCLUSIONS

The static and dynamic performance of an elevation drive system for the ARES 90mm gun will be strongly affected by the weapon's large unbalance and the characteristics of the components used in the gun drive system. One of the components that will be required is an equilibrator if military standards with respect to manual control forces and rates are to be met. Although an equilibrator will have a significant impact on manual gun control, its effect on stabilization performance will be small.

Stabilization of the ARES 90mm gun will require a more powerful drive system than would be the case if the gun were balanced. In this study, the interrelationship of gun unbalance, drive system parameters, and positioning accuracy have been defined. A comparison of the impact of unbalance on electric and hydraulic gun drives shows the hydraulic drive is less affected, due to its high torque gain. The better performance and relatively small size and weight of a hydraulic actuator resulted in its selection as part of a stabilization servo that was analyzed in depth.

The analysis defined the expected variation in gun pointing angle when on the move with a high performance electrohydraulic servo for three configurations:

1. A dynamically balanced gun with the inertia of the ARES 90mm gun
2. An unbalanced ARES 90mm gun driven by the actuator selected for a balanced gun
3. An unbalanced gun with an actuator having twice the area (i.e., twice the force capability) of that used for the balanced gun.

Graphs are presented that define the stabilization performance as a function of unbalance and actuator size as well as the performance with coincident firing. The estimates of servo performance were obtained from a linear digital computer model developed for analysis of the XM-803, XM-1, and DIVAD elevation servos.

The results of this analysis indicate that the 0.3 to 0.4 mil rms gun stabilization accuracy required for on-the-move anti-armor systems is achievable for the unbalanced 90mm gun. A 3,000-pound per square inch (absolute) hydraulic system with an actuator area of 9 to 10 square inches can be configured to achieve this performance. Although this actuator has twice the area that would be required for a similar balanced gun, it is smaller than actuators used for heavy battle tanks. The actuator and other components that would be reuired to stabilize the 90mm gun are readily available.

SECTION III

GUN AND VEHICLE CHARACTERISTICS

To estimate gun stabilization system errors, critical vehicle and gun characteristics must be defined. For this study, the vehicle was assumed to be a rigid body having mass and inertia properties specified in TARADCOM "Concept 22." This configuration is shown in Figure 3-1. Specific parameters for the vehicle and the gun are contained in Tables 3-1 and 3-2.

Since human factors place an upper bound on vehicle base motions, gun stabilization systems on light weight armored vehicles will experience terrain-induced disturbances similar to those that have been measured on existing vehicles. Consequently, the turret pitch rate and normal acceleration used as inputs in this analysis were derived from measurements on the HIMAG, General Motors XM-1, and XM-803 vehicles. This data represents an upper bound for the motion experienced while traversing a variety of terrains at difference speeds. (See Figure 3-2.)

Aside from the large unbalance, the 90mm gun parameters are not significantly different from the values for guns that have been stabilized on other vehicles. Although the "Concept 22" vehicle is relatively light, its inertia is large enough that reaction torques when elevating the weapon will not impact performance. When designing a turret drive for this vehicle, however, the reaction torques will have to be considered.

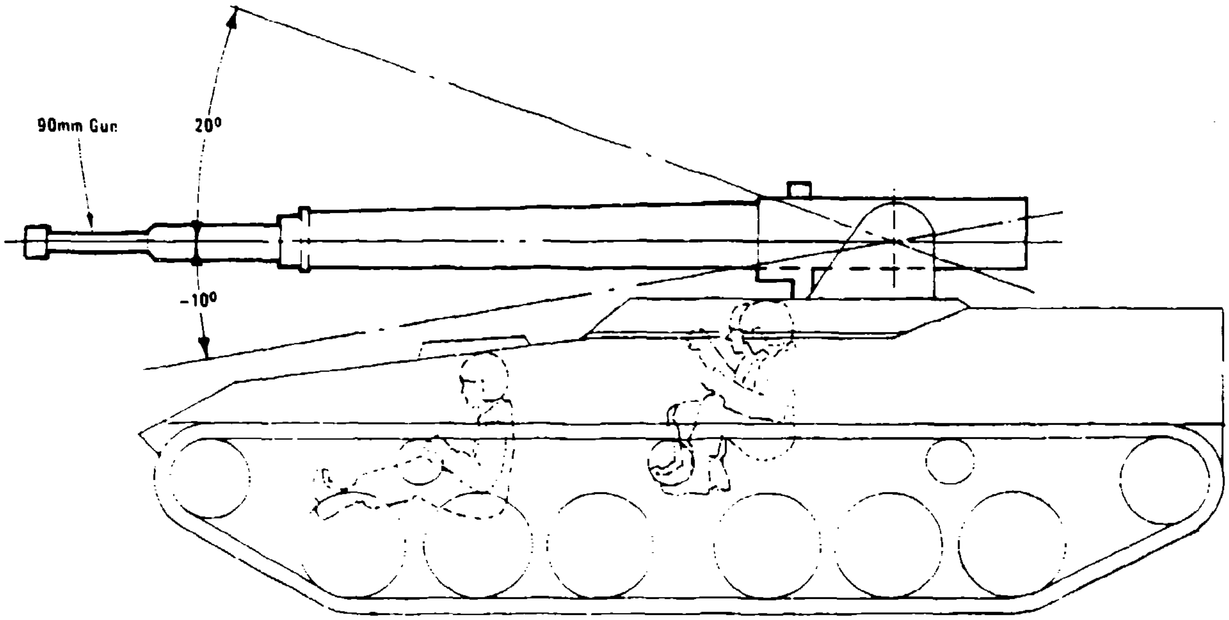


Figure 3-1. TARADCOM Concept 22

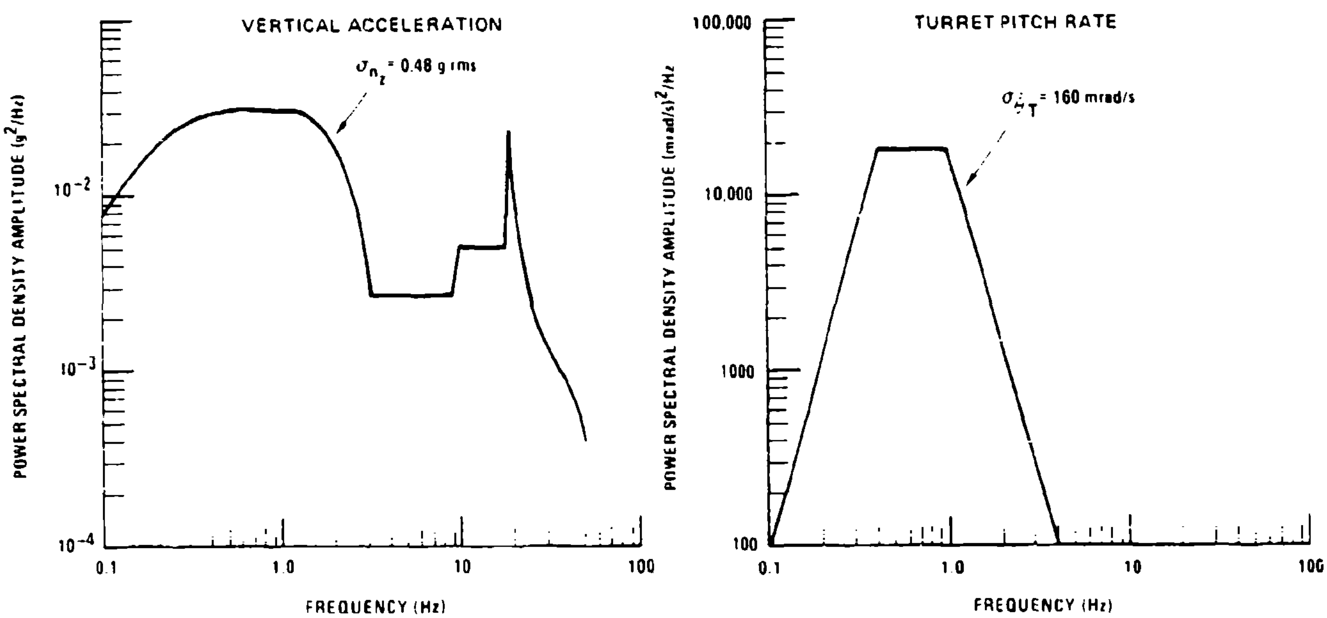


Figure 3-2. Disturbance Input Power Spectral Densities

Vehicle Combat Weight		41,855 lb
Ground Pressure at Combat Weight		8 lbf/in^2(a)
Physical Dimensions:		
• Length		246 in
• Width		110 in
• Height		96 in
• Ground Clearance		19 in
• Wheel Base		150 in
Suspension:		
• Type, Hybrid		--
• Wheel Travel (Static to Bump Stop)		14 in
Road Wheel Diameter		24 in
Track Width		17-1/2 in
Gross Power/Weight		24 hp/ton
Fuel Capacity		170 gal
Cruising Range		300 mi
Suspension Dynamics		
• Undamped Natural Frequency		68 cpm
• Damping Ratio, Jounce		0.6 C_c
• Damping Ratio, Rebound		0.3 C_c
Suspension Stiffness		
• Vertical Static Spring Rate		412.4 lb/wheel
Engine		
• Make	Cummings	
• Model	VIA-903	
• Power	500 hp	
Power Train:		
• Make	Allison	
• Model	X-300	
Armament:		
• Primary	90mm	
• Secondary	0.50 cal	
• Supplementary	7.62mm	
Ammunition Load:		
• Primary	50	
• Secondary	1,000	
• Supplementary	1.600	

Table 3-1. Concept 22 Vehicle General Data

ITEM	% GVW	WEIGHT (kg)	WEIGHT (lb)
Hull	22.2	4,214	9,290
Suspension	21.4	4,105	9,050
Power Train	16.1	3,062	6,750
Aux Auto	3.0	570	1,255
Weapon Station	23.4	4,445	9,800
Ammo	6.5	1,228	2,710
Crew	2.7	500	1,100
OVE	1.7	317	700
Fuel	3.0	544	1,200
	100.0	18,985	41,855

CENTER OF GRAVITY AT GVW	DISTANCE (mm)	DISTANCE (in)
Longitudinal from Front of Hull	3,225	127
Vertical from Ground	1,170	46
Lateral from Centerline of Vehicle	50 (Right)	2

INERTIAS	X_X	X_Y	X_Z	UNITS
Sprung Mass	8,386	34,078	32,378	slug-ft^2
Nonrecoiling parts of gun assembly about trunion axis	29	138	142	slug-ft^2
Recoiling parts about their center of gravity	74	1,636	1,100	slug-ft^2

GUN PARAMETERS	CANNON ONLY	CANNON PLUS EMPTY FEEDER	CANNON PLUS 18 ROUNDS IN FEEDER	UNITS
Weight	2,964	3,414	4,584	lb
Imbalance	5,242	4,717	2,474	ft-lb
Inertia about trunion	1,206	1,268	1,573	slug-ft^2
Distance from center of gravity to trunion	21.2	16.6	6.5	in
First bending mode	—	14	--	Hz

Table 3-2. Concept 22 Weight and Inertia Data

SECTION IV
DESIGN CONSIDERATIONS IN COMPONENT SELECTION

4.1 ELECTRIC VERSUS HYDRAULIC DRIVES

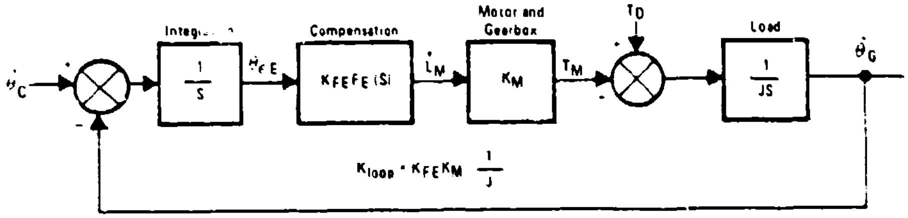
Successful gun positioning systems for armored vehicles have been built using either electric or hydraulic drive components. However, the positional change due to disturbing torques, such as those due to gun unbalance, is generally higher for electric drives than hydraulic. As will be explained in the following paragraphs, the relative sensitivity of electric and hydraulic servomechanisms to external torques is a direct result of fundamental properties of electric motors and hydraulic actuators. Consequently, the systems with electric drives generally have larger errors than systems with hydraulic drives.

The relationship of position error (θ_{EE}, θ_{EH}) to disturbing torque (T_D) for electric and hydraulic servos can be determined from the block diagrams presented in Figure 4-1. For a constant disturbing torque and zero command rate $F_E(S) = 1.0$, and the relationship of position error to torque for the electric drive is

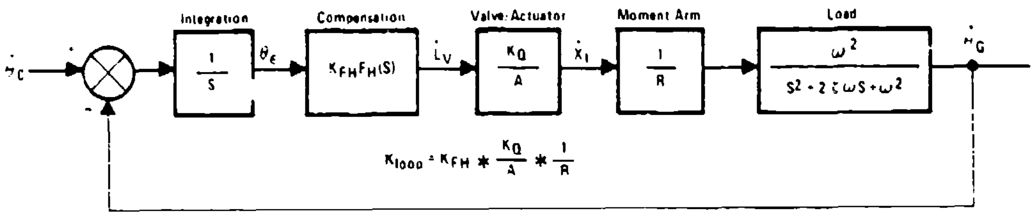
$$\theta_{EE} = - \frac{T_D}{K_{FE} K_M} ,$$

where $K_{FE}K_M$ equals the forward path dc gain. To obtain a numerical value for θ_{EE}, values have to be defined for T_D and $K_{FE}K_M$. An estimate of $K_{FE}K_M$ can be obtained from the product of the load inertia (J) and the loop gain (K_{loop}).

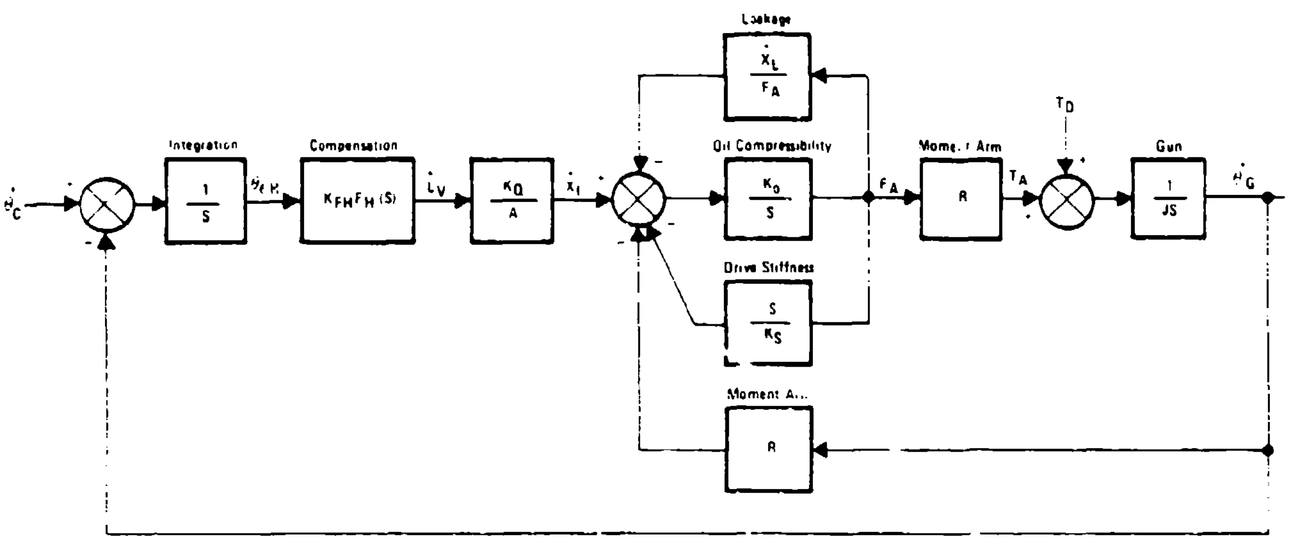
Since the gearbox should equal less than 20 percent of the load inertia, we can approximate J by the gun inertia (1,268 slug-ft^2). An estimate of the loop gain can be made by recognizing that the electric drive is a Type 2 system (one integration from rate error to position error, the other integration in the load from torque to rate), with a load that resonates on the stiffness of the drive linkages. From previous experience it is known that structural limitations will result in a fundamental load resonance below 15 hertz. With the



(a) Electric Gun Servo, Simplified



(b) Linear Hydraulic Gun Servo, Simplified



(c) Linear Hydraulic Gun Servo

Figure 4-1. Electronic Versus Hydraulic Drive Comparisons

gain variations as a function of frequency for the compensation assumed to be minor effects, the stability constraint will require the loop gain divided by the frequency squared to be less than unity at 15 hertz, or

$$K_{loop} \leq 8{,}870 \text{ s}^{-2}$$

With this value of loop gain and the total unbalance of the 90mm gun supported by the drive system, the error would be:

$$\theta = -\frac{T_D}{K_{FE}\, K_M} = -\frac{T_D}{J\, K_{loop}} = \frac{4{,}717 \text{ ft-lb}}{(1{,}268 \text{ ft-lb-s}^2)\ (8{,}870 \text{ s}^{-2})} = 0.4 \text{ mrad}$$

The magnitude of this static error can be reduced by static equilibration of the load. However, the fundamental interrelationship of position errors, loop gain, and disturbing torques will still exist for dynamic torques such as those that occur while traversing rough terrain.

As shown in Figure 4-1b, the hydraulic servo is a Type 1 system, that is, a single integration occurs inside the loop. Consequently, position error and disturbing torques are not related in the same manner as for electric drives. To determine the effect of an external torque on the hydraulic drive, more of the drive system parameters have to be defined as shown in Figure 4-1c.

In Figure 4-1c the internal leakage has been shown. Even through small enough to be neglected for dynamic conditions, it has a major effect on the static torque gain of the servo. In fact, the static error is linearly related to the leakage by the following relationship, which can be derived from the block diagram.

$$\theta_{\epsilon H} = \frac{1}{K_{FH}} * \frac{A}{K_Q} = * \frac{\dot{X}_L}{F_A} * \frac{1}{R} \quad T_D$$

where

K_{FH} = gain between position error and valve current

K_Q = valve flow gain

A = actuator area

R = moment arm

$\dot{X}_L/F_A$ = rate of change of actuator position due to leakage.

As with the electric drive, stability considerations limit the loop gain and the forward-path gain. The relationship between loop gain and forward-path gain for the hydraulic servo is

$$K_{FH} * \frac{K_Q}{A} = R * K_{loop}$$

The error can then be expressed by

$$\theta_{EH} = \frac{T_D}{K_{loop} * R^2} * \frac{\dot{X}_L}{F_A}$$

Some representative values for the indicated parameters are

$K_{loop} = 200\ s^{-1}$

R = 15 in

$$\frac{\dot{X}_L}{F_A} = 1.5 * 10^{-5}\ \frac{in/s}{lb}$$

The error to support the total gun unbalance would then

$$\theta_{\epsilon H} = 4{,}717\ ft\text{-}lb * 12\ \frac{in}{ft} * 1.5 * 10^{-5}\ \frac{in/s}{lb} = 0.02\ mrad$$

This is a factor of 20 lower than for the electric drive, and it results from the error being a function of the force-to-rate relationship $\dot{X}_L/F_A$ as well as the loop gain. For the electric drive, the error primarily depends upon the loop gain, which is limited by stability considerations.

Under dynamic conditions, the effect of leakage is negligible, and the error in a hydraulic servo due to external torques such as those caused by an unbalanced gun is a function of the compressibility of the oil in the actuator and the loop gain. As a result of the high bulk modulus for hydraulic fluids, the errors in a hydraulic servo will be less than those in an electric servo with an unbalanced load. The calculation of these errors requires a detailed servoanalysis for both electric and hydraulic powered systems.

In summary, it can be stated that hydraulic drives have a performance advantage over electric drives when positioning unbalanced, high inertia loads as a result of their high torque gain. This high torque gain depends upon characteristics (low leakage and high fluid bulk modulus) that are independent of stability considerations that, in turn, limit high loop gains. The net result of the higher torque gain is smaller position errors.

In addition to their better positioning capability, hydraulic gun drives generally weigh less and require less space than comparable electric drives when hydraulic power is already available or can be easily obtained. Consequently, it was decided that the appropriate drive system for analysis in this feasibility study would be a hydraulic drive. Since a linear hydraulic actuator is simpler and significantly less expensive than a hydraulic motor/gearbox drive yet can provide the desired angular freedom, it was the concept selected for more detailed analysis.

4.2 SERVOACTUATOR CHARACTERISTICS

Weight and size are critical concerns for the class of combat vehicle being addressed here. Consequently, it is important to understand the parameters that will have a significant impact on the size and weight of the stabilization system. In a hydraulic servo, the primary weight and volume requirements are established by the servovalve and actuator. Consequently, the following paragraphs describe the considerations that determine the specification of these components.

The primary determinant of size for a hydraulic actuator to be used in a gun servo is its stiffness. This results from the impact of this stiffness on system dynamics. As a first approximation to the load dynamics the gun, mount and actuator can be represented as shown in Figure 4-2. In this model, the inertia for the 90mm gun assembly (15,216 in-lb-s^2) has been separated into two pieces: the gun barrel inertia, J_G, (10,144 in-lb-s^2) and the mount inertia, J_M, (5,072 in-lb-s^2). The gun barrel is assumed to be constrained by the stiffness of the mount (K_G) and a damping between the gun and the mount (B_G). For this analysis, the stiffness (K_G) was set at 78.5 * 10^5 inch-pounds per radian to produce a resonant frequency equal to the value provided by ARES (14 Hz). The damping was set at a value (10^{10} in-lb-s/rad) that represents values measured for similar guns used on other vehicles.

The actuator is represented as a simple spring (K) whose magnitude depends upon the bulk modulus of the fluid (B), the area of the actuator (A), and the entrapped volume (V) as shown in the following equation

$$K = \frac{4BA^2}{V}$$

For frequencies below 14 hertz, the gun assembly is effectively an inertia (J_M + J_G) supported by the actuator spring (K) acting through a radius (D_1). These parameters establish the resonant frequency (ω) in the simplified block diagram of a Linear Hydraulic Gun Servo in Figure 4-1b. Since the loop gain and bandwidth for the servo (and consequently its performance) are directly related to the load resonant frequency, it is desirable to make this frequency as high as possible.

From experience on previous combat vehicle gun systems, Delco has established that sufficient stiffness can be obtained in reasonably sized actuators to produce a fundamental load resonant frequency of at least 63 radians per second, which is compatible with the desired stabilization performance. Consequently, this value was selected as a parameter for this study.

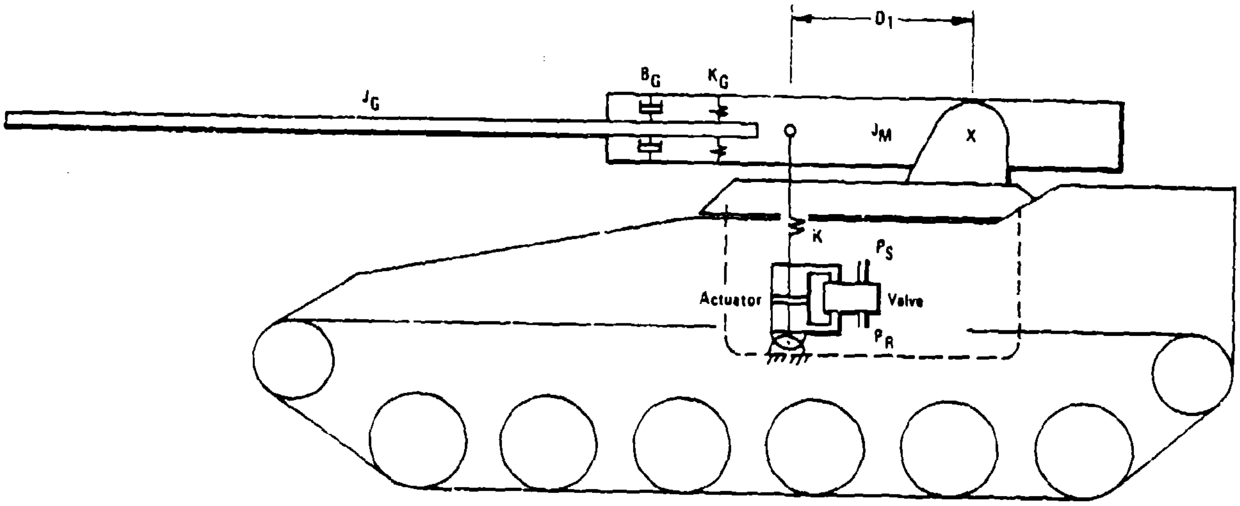


Figure 4-2. Elevation Gun Drive System Model

Since the load resonant frequency is a product of the actuator area (A) and moment arm (D_1), a designer can decrease one of these parameters while increasing the other to obtain the desired result. However, large values of D_1 will also result in large actuator strokes to rotate the gun through the desired angle. Consequently, selection of an actuator requires tradeoffs among stroke, moment arm, and area to achieve the objectives. For the current study, baseline values were selected after consideration of component sizes that had been used successfully on previous programs as shown below.

PROGRAM	MOMENT ARM	AREA	STROKE
XM-803	11 in	16.87 in^2	6.25 in
XM-1	15 in	12.89 in^2	8.25 in

The parameters selected were: moment arm 14.3 inches, actuator area 4.7 in^2, and actuator stroke 7.5 inches. An actuator with these characteristics will be capable of producing gun accelerations greater than 3 radians per second squared, which represents a reasonable value for systems of this type.

In addition to the parameters that have been defined, the size and weight of the gun drive system components are a function of the hydraulic supply pressure. As shown in Figures 4-3* and 4-4*, there is an optimum supply pressure between 3,000 and 4,000 pounds force per square inch (absolute) for actuators with strokes (X_M) from 1 to 12 inches, no-load velocities ($\dot{X}_M$) from 1 to 10 inches per second, and stall forces (F_M) from 100 to 100,000 pounds. Since the actuator selected for this study has parameters close to those for the L3 actuator in these curves, it will exhibit similar properties. Since 3,000 pounds force per square inch (absolute) represents a practical limit for many currently available components, it was selected as the baseline supply pressure for this system.

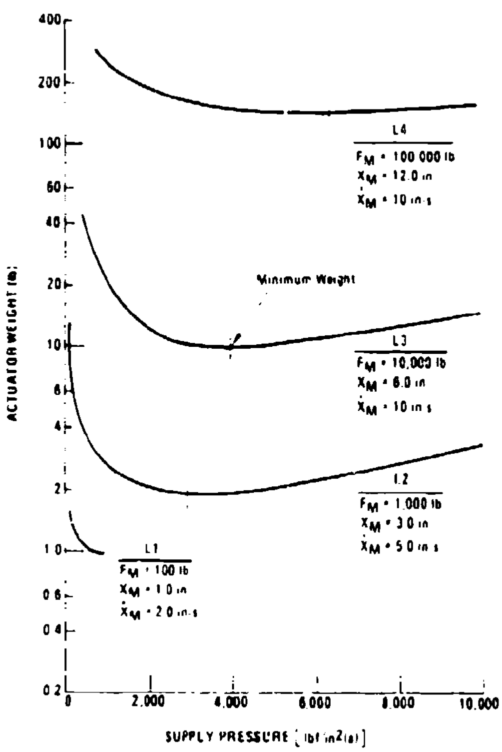


Figure 4-3. Actuator Weight Predictions

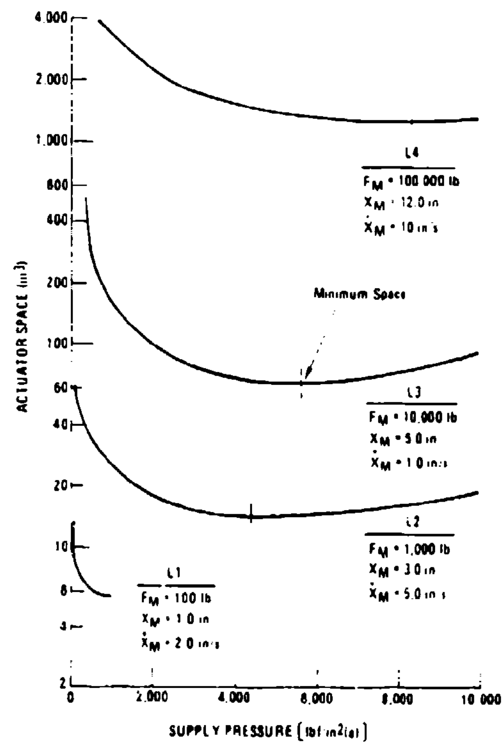


Figure 4-4. Actuator Space Predictions

*W.J. Thayer, Supply Pressure Considerations for Servoactuators, MOOG Technical Bulletin 119.

4.3 EQUILIBRATION

The primary concern in this study is gun stabilization, that is, gun position control when the vehicle is moving. However, there are also significant static effects when positioning an unbalanced gun, which are summarized in the following paragraphs.

The large gun unbalance discussed in Section III requires that a drive system product a significant torque merely to hold the weapon in place or rotate it in a direction opposing gravity. Although the power drives can be designed to produce the required torque, it is not possible to design a conventional manual drive, generally a backup mode, to produce both the required torques and reasonable gun rates. The desired limit for the elevation manual control force* is less than 5 pounds, and the desired gear ratio* is 10 to 15 mils of gun elevation per handle revolution. Meeting these requirements with a 100 percent efficient drive and a manual control radius of 1 foot would produce 1,800 foot-pounds of torque at the gun. This is substantially less than the 4,717 foot-pounds of gun unbalance, even without considering realistic efficiency of the drive and handle radii. Consequently, some form of equilibration will be required if the ARES 90mm gun is to be elevated manually by conventional techniques.

When equilibrators are added to a gun, they can affect the stabilization servo performance. As shown in Figure 4-5, the external torque acting on the servo equals the torque produced by the acceleration acting on the unbalanced mass minus the torque produced produced by the equilibrator. For a system with a carefully designed equilibrator, the output torque will be only a function of gun position and independent of velocity and acceleration. In this case, the static torques should cancel, leaving an external torque equal to the product of the mass unbalance and the variations in vertical acceleration.

*Human Factors Engineering Design for Army Material, MIL-HDBK-759.
Human Factors Design Standard for Vehicle Fighting Compartments, Human Engineering Laboratories, Standard, S-264A.

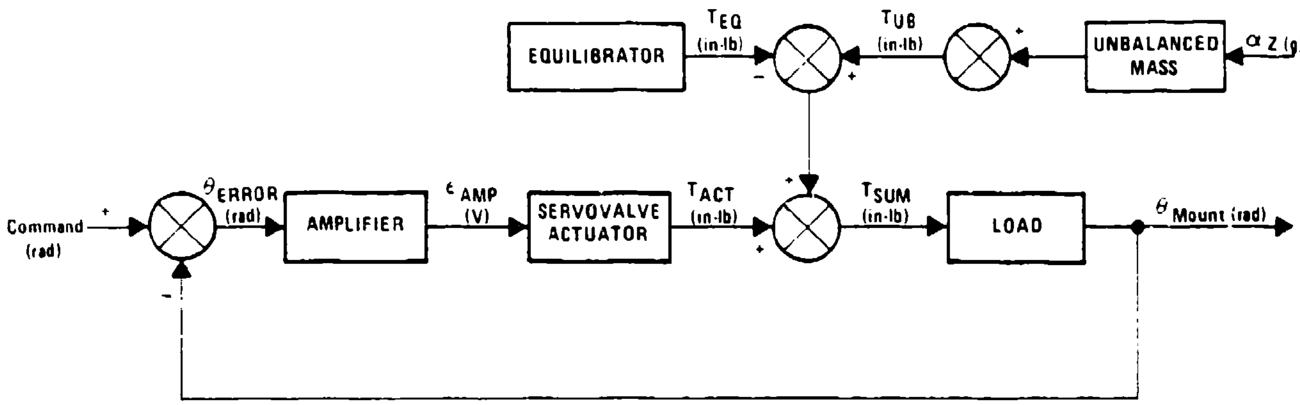


Figure 4-5. Elevation Model With Equilibrator, Simplified

In designing the equilibrator, consideration should be given to the variation in torque with elevation angle and the dynamic properties of the equilibrator. Since the power mode can excite frequencies that would not have significant energy during manual operation, an equilibrator for this system can not be designed purely as a static device. Careful attention must also be given to the damping on mechanical resonances.

To evaluate the effect of the equilibrator on performance in the stabilized mode, the pointing error for this system without an equilibrator was calculated. Several simplifying assumptions were required to make this analysis manageable:

1. System nonlinearities (actuator friction, bearing clearance, etc.) are neglected.
2. Damping of the mount is small enough to be neglected.
3. Resonant frequencies in the electronic compensation networks are much greater than the significant excitation frequencies.
4. Force produced by the equilibrator is independent of the rate of motion and position of the weapon.

These are reasonable, practical assumptions for high performance electrohydraulic servomechanisms.

If a servovalve were ideal (no leakage), no power would be required to maintain the load position under the influence of steady external torques, and the error would be zero. However, as shown in Paragraph 4.1, practical devices have an error proportional to the magnitude of the static torque. This error will, of course, be reduced for any lowering of the static torque as a result of equilibration. Since the error calculated for the hydraulic drive in Paragraph 4.1 was small (0.02 mrad), there will be negligible practical benefit in the stabiized mode of having an equilibrator when the vehicle is stationary.

When mobile, an equilibrator decreases the dynamic pointing error for a hydraulic servo. The variation in dynamic pointing error for an unbalanced gun is caused by changes in the valve flow gain as a function of the pressure across the actuator. As actuator pressure increases due to external torques, the valve flow gain decreases, causing a corresponding decrease in the velocity constant for the servo. This results in a larger error (θ_ϵ) for the same input to the servo. An approximation to the change in flow gain can be made ' assuming a low pressure required to move the load in response to exte..al inputs. The ratio of the flow gain while supporting an unbalance to the flow gain without an unbalance for the ARES 90mm gun servo is

$$\frac{K_{QUB}}{K_{QNOM}} = \frac{P_S - P_{ACTUB}}{P_S} + \frac{P_S \; T_{UB/A}R}{P_S} = \frac{(3{,}000 - 840)}{3{,}000} = 0.85$$

where

K_{QUB} = Valve flow gain with an unbalance

$K_{Q\ NOM}$ = Valve flow gain with balance

P_S = Supply pressure [3,000 lbf/in^2(a)]

P_{ACTUB} = Acutator pressure due to unbalance [lbf/in^2(a)]

Small changes in gain, such as the 15 percent variation calculated here, can have a significant impact on high performance servos that use open loop (base motion decoupling) techniques in addition to the attenuation of the closed loop. The change in stabilization as a result of gain variations caused by actuator differential pressures such as those due to gun unbalances is presented in Figure 4-6. Since the effect of the equilibrator would be to remove

the static unbalance that causes the gain variation, installing the equilibrator would remove this error source. However, if an equilibrator is not installed, the amount of error due to this effect can also be reduced by increasing the size of the actuator and changing the moment arm through which its works.

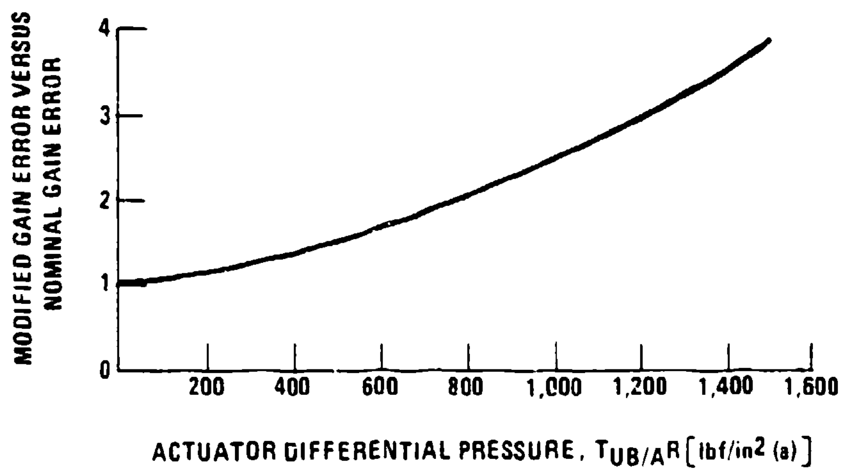


Figure 4-6. Servo Dynamic Error Due to Unbalance

In summary, although an equilibrator is only a static device, it can alter both the static and dynamic performance of the stabilization system. It does this by changing the steady state torque level in the system, which alters the pressure across the actuator. This pressure directly affects the error that exists in the servo, both statically and dynamically.

SECTION V
LINEAR MODEL DEVELOPMENT

5.1 SERVO DESIGN

To calculate gun pointing accuracy when traversing rough terrain, a system mechanization has to be defined. The approach used in this analysis follows what Delco has used successfully in other gun elevation servos. As shown in Figure 5-1, the basic servo controls gun rate in response to commanded rate. It uses a signal proportional to actuator differential pressure to provide artificial load damping and electronic compensation to acheive high loop gain and reasonable stability margins. In addition, open loop compensation, where turret pitch rate is measured and used to command motion of the actuator, aids in decoupling the gun from the base motion.

The artificial damping and electronic compensation in this servo are selected to achieve an outer loop gain of 200 seconds^{-1}, with a gain margin greater than 6 decibels and a phase margin greater than 40 degrees. These gain and phase margins allow reasonable variations in component parameters (valve flow gains, actuator friction, gun system resonant frequencies, and compensation network characteristics), with the performance of the servo being relatively unaffected. The outer loop gain of 200 seconds^{-1} will produce accurate following of steady state rate commands (error <0.1 mrad). Analysis of the servo to determine the required compensation parameters was performed with the model described in the next section.

5.2 ESTIMATION OF STABILIZATION PERFORMANCE

Analysis of the gun servo to determine the compensation parameters and calculation of the stabilization performance was accomplished with the mathematical model shown in Figure 5-2. The hydromechanical portions of this diagram are the same as described in Paragraph 4.1, with the exception of the valve dynamics, which have been incorporated here because they impact the compensation design. The remaining blocks show the dynamics for the turret gyro, which senses pitch rate and the electronic compensation. The turret gyro dynamics

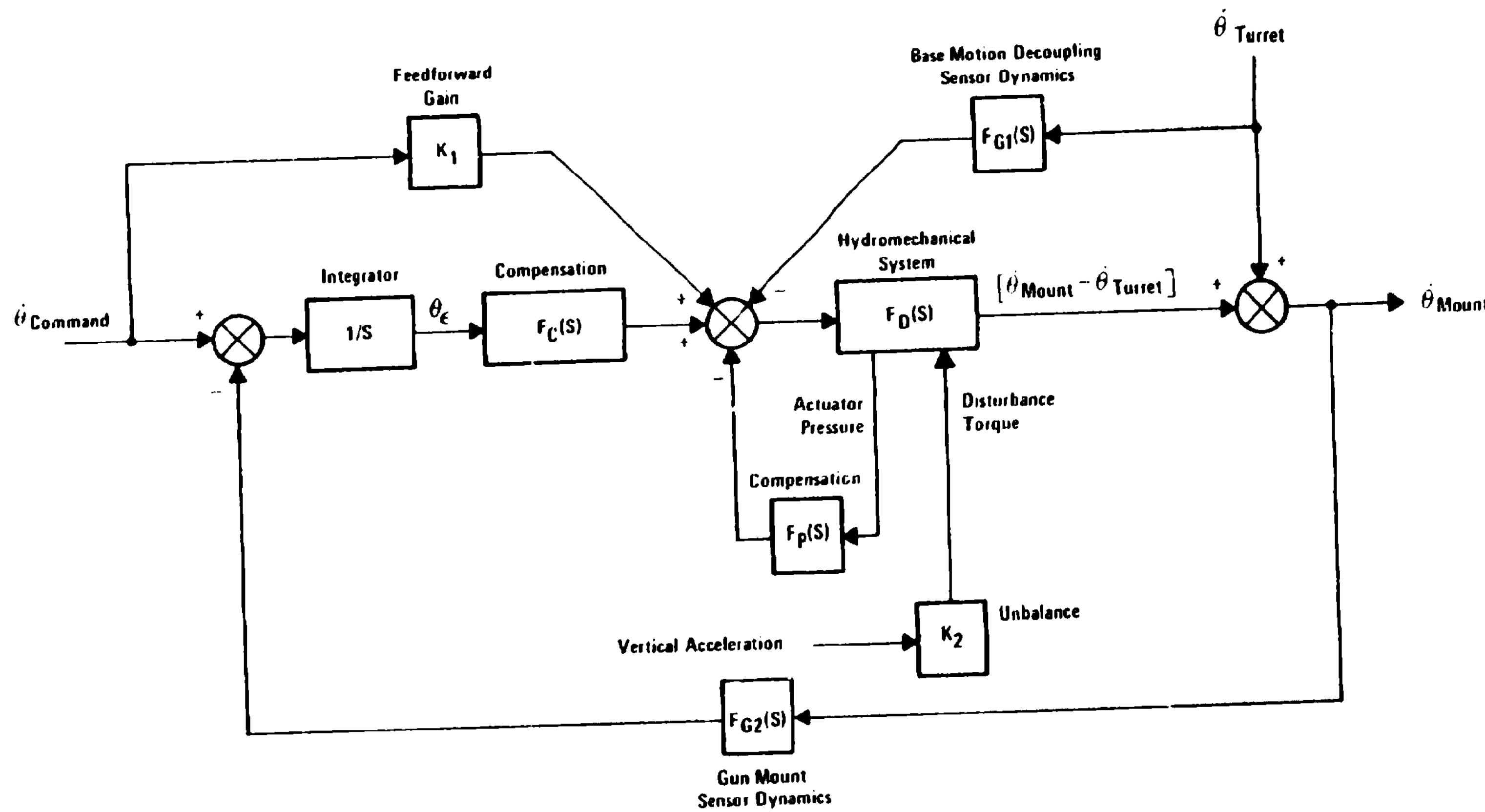


Figure 5-1. Gun Elevation Servo Block Diagram

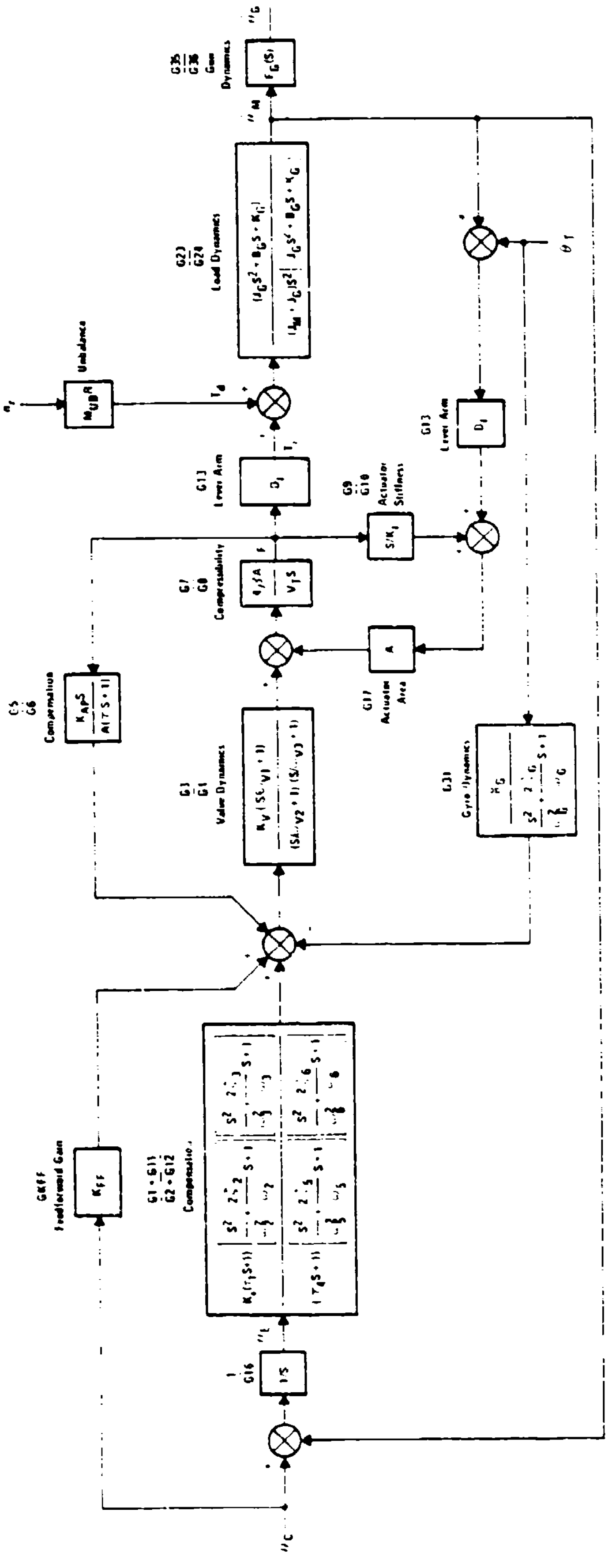


Figure 5-2. Gun Control System Dynamic Model

are generally lower frequency and impact system performance more than the gun mount rate gyro dynamics, which have been deleted.

Electronic compensation occurs in the force feedback path (that is, artificial damping derived from actuator pressure) and in the forward path after the error signal (θ_ε). The compensation shown in the force feedback path is a simple high pass filter that is required for high static stiffness. The parameters in the forward path compensation were selected such that, when combined with the response for the rest of the system, the desired stability margins were achieved. Numerical values for these parameters and those in the hydromechanical portions of the system are given in Table 5-1.

Calculation of gun servo position errors when traversing rough terrain requires definition of turret angular rates and translational accelerations and the relationship between these inputs and the gun position. Estimates of turret pitch rate and vertical acceleration for the vehicle on which the ARES 90mm gun will be mounted were defined in Section III. Since these motions are random, their definitions are in the form of power spectral densities. The change in gun position as a result of a random input is calculated from

$$\delta^2 = \int_0^\infty \Phi(\omega)\,|T(j\omega)|^2 d\omega$$

where

δ^2 = variance of servo position

$\Phi(\omega)_j$ = power spectral density of disturbance

$|T(j\omega)|$ = magnitude ot the transfer function between the disturbance input and the servo error.

```
GHSF = 12.5
GKBC = 10.0
GKFF = 12.5
G1 = 2717*(S/20.11 + 1)*(S/31.42)**2 + 0.80*S/31.42 + 1)
G2 = (S/3.88 + 1)*((s/62.83)**2 + 1.40*S/62.83 + 1)
G3 = 40.*1.12*284.7*(S/81.2+1.)*(4.714/12.38)
G4 = (.442*S*(S/16.2+1)+20.*40.*1.12*(S/81.2+1.))
G5 = .00025*S
G6 = (S/5.10+1)*4.714
G7 = 4*140000*4.714
G8 = 42.44*S
G9 = S
G10 = 4.9E6
G11 = ((S/175.9)**2 + .15*S/175.9 + 1)
G12 = ((S/175.9)**2 + .90*S/175.9 + 1)
G13 = 14.313
G16 = S
G17 = 4.714
GJM = 5078
GJG = 10144.
GBG = 29.231E3
GKG = 78.492E6
GBB = 1.E10
GKB = 1.E10
G23 = GJB*GJC*S**4 + (GJB*GBG + GBG*GBS + GBG*GBB)*S**3 +
1 (GJP*GKG + GJG*GKG + GJG*GKB + GBG*GBB)*S**2 + (GBB*GKG + GKB*GGBG
2)*S + GKB*GKG
G24 = GJM*GJG*GJB*S**5 + (GJM*GBG*(GJB + GBG) + GJG*GBB*(GJM + GJB
1))*S**4 + (GJM*GKG*(GJB + GJG) + GBB*GBG*(GJM + GJB + GJG) +
2 GKB*GJG*(GJM + GJB))*S**3 = (GJM+GJB + GJG)*(GBB*GKG + GKB*GBG)
3 *S*E2 + GKB*GKG*(GJM + GJB + GJG)*S
GBA = 2500
G24 = G24 + G23*GBA

TURRET PITCH BUCKING COMPENSATION
G31 = GKBC/((s/188)8**2 + 1.4*S/188 + 1
G33 = 14.313
```

Table 5-1. Elevation Gun Servo Parameters

Assuming the turret pitch rate and normal acceleration will not be strongly correlated on the vehicle, the root-mean-square value of the gun position will be

$$\delta_{TOTAL} = \delta^2 E/_{\theta} + \delta^2 E/nz$$

where

δ_{TOTAL} = rms value of gun elevation position while traversing rough terrain

$\delta^2 E/_{\theta}$ = position variance due to turret pitch rate

$\delta^2 E/nz$ = position variance due to turret normal acceleration

The transfer functions required in these expressions were calculated from the model previously described.

SECTION VI
PERFORMANCE ESTIMATES

6.1 PERFORMANCE OF THE BASELINE SYSTEM

When the gun and vehicle characteristics from Section III are combined with the mathematical relationships described in Section V, the expected variation in gun position due to the vehicle traversing rough terrain can be calculated. The result is an error of 0.6 milliradian rms. If this error were directly translatable into firing accuracy, it would represent unacceptable performance for anti-armor applications. The amount of this error attributable to the gun unbalance can be found by setting the unbalance equal to zero in the model. This calculation yields a value for the error of 0.3 milliradian rms.

Since the baseline error due to the weapon's unbalance was significant, additional analyses were conducted to determine what stabilization performance improvement could be achieved using techniques proven successsful on previous programs. The two techniques offering the most promise are increased system stiffness through the use of a larger actuator and reduction of the errors at the time of firing by means of coincident firing. The performance impact of these techniques is described in the following paragraphs.

6.2 PERFORMANCE WITH LARGER ACTUATORS

As described in Section IV, the deflection of a hydraulic actuator due to external torques such as those caused by an unbalanced load impacted by accelerations is inversely related to the area of the actuator. Consequently, it is possible to increase system stiffness and reduce the servo errors by increasing the area of the actuator. To quantify the improvement that could be achieved with the 90mm gun servo, the variation in gun position was also calculated using the models described and an actuator with an area of 9.4 square inches, which is approximately twice the value used for the actuator in the baseline. The result was 0.4 milliradian rms, which represents a significant improvement. Since this performance can be achieved with an actuator that is

smaller than those used on the XM-1 and XM-803 vehicles, it is recommended that the impact of actuator size on performance be carefully considered when the servo for the 90mm gun is designed.

The results described above are for a specific unbalance (4,700 ft-lb), actuator areas (4.7 in^2 and 9.2 in^2), and moment arm (14.3 in). Since these are only representative values and will change as the 90mm gun program and vehicle concepts evolve, it is important to define the impact of variation in these parameters on stabilization performance. This data is presented in Figure 6-1, where stabilization error is plotted as a function of gun unbalance, actuator size, and moment arm. Letting the abscissa be a function of interrelated parameters results in single curve rather than a family of curves, which would be required if only unbalance were used for the abscissa.

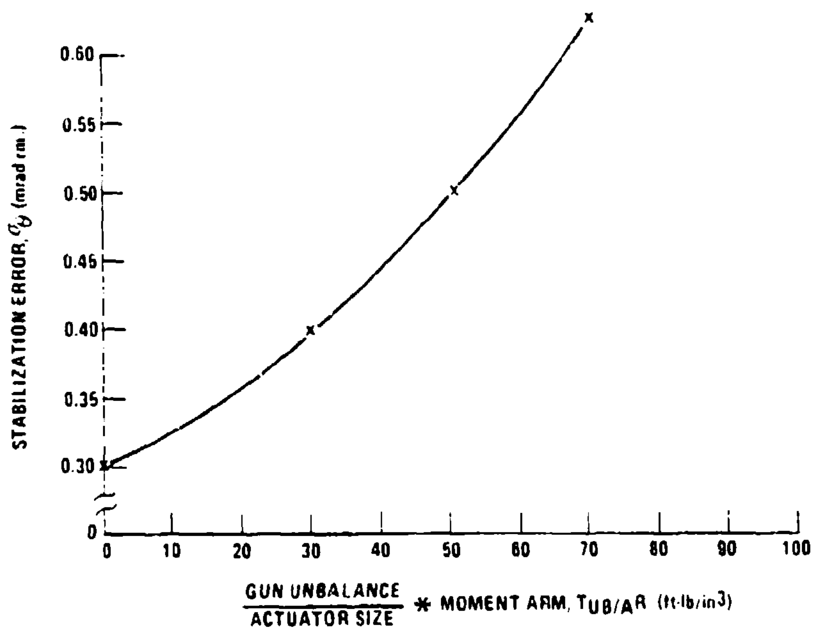


Figure 6-1. Stabilization Performance Data

6.3 PERFORMANCE WITH COINCIDENT FIRING

Although a significant performance improvement can be achieved with larger actuators as described in the preceeding paragraph, it may not be possible to

obtain the desired accuracy by this technique alone. Another technique that can be used to decrease the error at the time the gun is fired is coincident firing. When the coincident firing technique is mechanized, the firing circuit for the weapon is inhibited whenever servo errors exceed a specified value, called the Coincident Firing Limit (CFL). To evaluate the effect of this constraint on pointing accuracy requires a two-step process. The first is to generate time histories of the error signals for the fire control system with the simulated vehicle moving cross country. These time histories are then computer processed to simulate the firing procedure.

Figure 6-2 illustrates the statistical improvement in firing accuracy that can be expected using this technique in conjunction with a high performance servo. As shown, servos with a basic stabilization accuracy as high as 0.6 milliradian rms can be constrained to firing errors in the order of 0.3 milliradian rms, which is required for anti-armor applications. The expected time delay while waiting for coincidence to be achieved is in the order of 100 milliseconds, which, in a practical sense, has no effect on the perceived operation of the system.

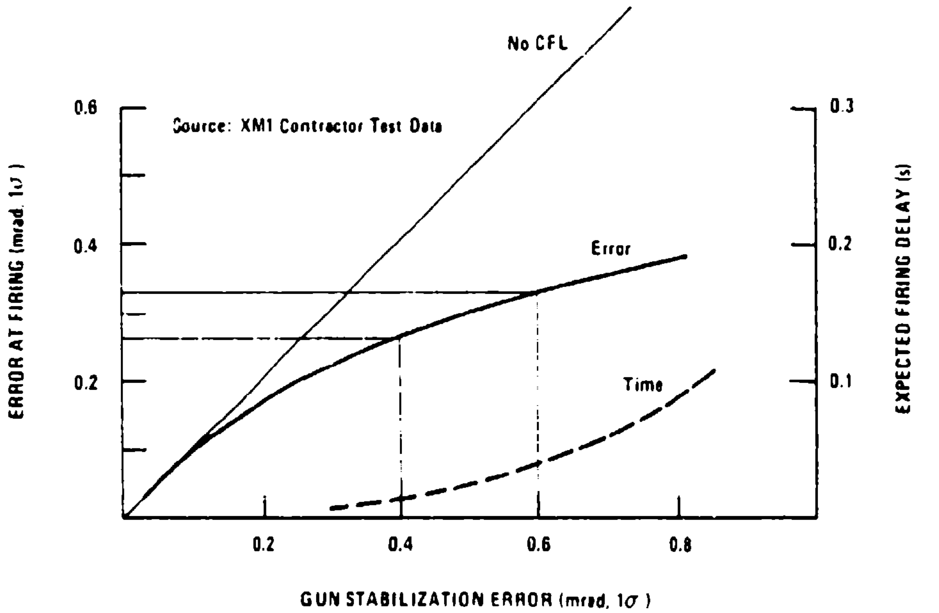


Figure 6-2. Error Compensation at Firing Due to Simple Coincident Limiter